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Surface Roughness Effect on the Performance of a Magnetic Fluid Based Porous Secant Shaped Slider Bearing

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Abstract

It has been sought to analyze the performance of a porous rough secant shaped slider bearing under the presence of a magnetic fluid lubricant. The concerned stochastically averaged Reynolds equation is solved to get the pressure distribution, which is, then used to derive the expression for the load carrying capacity. Consequently, the friction is also calculated. Computed values of dimensionless load carrying capacity and friction are displayed in graphical form. The figures are presented here tend to establish that the use of magnetic fluid as a lubricant increases the load carrying capacity and decreases the friction. However, the present study reveals that the negative effect of porosity and standard deviation can be reduced to a large extent by the positive effect of the magnetization parameters in the case of negatively skewed roughness. Further, this reduction becomes more evident when the negative variance is involved.

Key words: slider bearing, secant shape, magnetic fluid, load carrying capacity, friction

1. Introduction

The slider bearing is perhaps the simplest and most often encountered bearing. They are frequently associated with reciprocating motion. In cross section they may be flat, convex, concave or V-shaped. Because of the use of squeeze film slider bearings in clutch plates, automobile, transmission and domestic appliances many investigators (Prakash and Vij 1973, Bhat and Patel 1981), dealt with the problem of a squeeze film slider bearing. Slider bearing has been studied for various film shapes (Pinkus and Sternlicht 1961, Bagci and Singh 1983, Ajwaliya 1984, Hamrock 1994), as slider bearing is often used for supporting transverse loads.

The main resource of roughness comes out from wear and friction. Real bearing surfaces, however, have a more or less random distribution of roughness. Even the contamination of lubricant contributes to roughness through chemical degradation. The roughness appears to be random in character which does not seem to follow any particular structural pattern. The effect of surface roughness was discussed by many investigators (Mitchell 1950, Davis 1963, Burton 1963, Tzeng and Saibel 1967, Christensen and Tonder 1969(a), Christensen and Tonder 1969(b), Christensen and Tonder 1970, Tonder 1972, Berthe and Godet 1973). Christensen and Tonder (1969(a), 1969(b), 1970), proposed a comprehensive general analysis for both transverse as well as longitudinal surface roughness. Subsequently, this approach of Christensen and Tonder (1969(a), 1969(b), 1970), formed the basis of the analysis to study the effect of random roughness on the performance of the bearing system in a number of investigations (Ting 1975, Prakash and Tiwari 1982, 1983, Prajapati 1991, 1992 Guha 1993, Gupta and Deheri 1996, Andharia et al. 1997, 1999).

All the above investigations dealt with conventional lubricant. Oil based or other lubricating fluid base magnetic fluids can act as lubricants. The advantage of magnetic fluid as a lubricant over the conventional ones is that the former can be retained at the desired location by an external magnetic field. The magnetic fluid is prepared by suspending file magnetic grains coated with surfactants and dispersing it in non-conducting and magnetically passive solvents such as kerosene, hydrocarbons and fluorocarbons. When a magnetic field is applied to the magnetic fluid, each particle experiences a force due to the field gradient and move through the liquid imparting drag to causing it to flow. Hence with the proper applications of magnetic field the magnetic fluid can be made to adhere to any desired surface (Bhat 2003).

Verma (1986) and Agrwal (1986) studied the squeeze film performance by taking a magnetic fluid as a lubricant. The analysis of Bhat and Deheri (1991(a)), regarding the squeeze film performance of porous annular disk using a magnetic fluid lubricant suggested that the application of magnetic fluid lubricant enhanced the performance of squeeze film bearing system. Bhat and Deheri (1991(b)), modified and developed the method of Agrwal (1986) to study the performance of a magnetic fluid based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. It was concluded that the magnetic fluid lubricant unaltered the friction and shifted the centre of pressure towards the inlet. Also, Bhat and Deheri (1995), considered the hydrodynamic lubrication of a porous slider bearing and compared the performance by taking various geometrical shapes. Here, they observed that mostly the magnetic fluid lubricant shifted the centre of pressure towards the out let edge. Shah and Bhat (2003), extended the analysis of Ajwaliya (1984) and developed the mathematical model for analyzing the effect of slip velocity on a porous secant shaped slider bearing with ferro fluid lubricant using Jenkins model. Deheri et al. (2005), discussed the performance of transversely rough slider bearing with squeeze film formed by a magnetic fluid.

Here, it has been shown that the magnetic fluid lubricant plays an important role in reducing the negative effect of roughness. Thus, it is sought to study and analyze the performance of a magnetic fluid based secant shaped porous rough slider bearing.

2. Analysis

The geometry and configuration of the bearing system is given in Figure (1).The assumptions of usual hydrodynamics lubrication theory are taken in to consideration in the development of the analysis. The bearing surfaces are assumed to be transversely rough. The thickness $h(x)$ of the lubricant film is [Christensen and Tonder (1969(a), 1969(b), 1970)],

$$h(x) = \bar{h}(x) + h_s$$

Here $\bar{h}$ is the mean film thickness and h_s is the deviation from the mean film thickness characterizing the random roughness of the bearing surfaces. h_s is assumed to be stochastic in nature and governed by the probability density function $F(h_s)$, which is defined by

$$F(h_s) = \begin{cases} \frac{32}{35b}\left(1 - \frac{h_s^2}{b^2}\right)^3 & -b \le h_s \le b \\ 0 & , \quad \text{other wise} \end{cases}$$

where b is the maximum deviation from the mean film thickness. The mean α, standard deviation σ and the parameter ε, which is the measure of symmetry of the random variable h_s , are defined by the relationships

$$\alpha = E(h_s),$$

$$\sigma^2 = E[(h_s - \alpha)^2],$$

$$\varepsilon = E[(h_s - \alpha)^3]$$

where, E denotes the expected value defined by;

$$E\,(R) = \int_{-b}^{b} Rf\,(h_s)dh_s$$

For secant shaped slider bearing:

$$h = h_0 \sec\left(\frac{\pi x}{L}\right)\;; \qquad\qquad h = \overline{h_0}\sec\left(\frac{\pi \bar{x}}{L}\right)$$

The shape is symmetrical about the origin [*x=0*, consider only the left hand side to start with].The surface extends from $x = -\infty$ to $x=0$. Now the runner must drag oil in to the wedge to produce a positive oil pressure, so that velocity is $-U$ (Cameron 1972). The magnetic field is oblique to the stator and its magnitude is given by

$$H^2 = kL^2 \sin\left(\frac{\pi x}{L}\right)$$

Where $k = 10^{14}A^2m^{-4}$ chosen so as to hence a magnetic field of strength over 10^5(Verma 1986, Bhat and Deheri 1991(a,b), Prajapati 1995).

Under the usual assumptions of the hydrodynamic lubrication gives the governing Reynolds equation as (Andharia et al. 1997, 1999).

$$\frac{\mathrm{d}}{\mathrm{dx}}\left(P - \frac{\mu_0\bar{\mu}H^2}{2}\right) = 6\mathrm{U}\eta\frac{h - \bar{h}}{\mathrm{A}(h)}$$

where

$$\mathrm{A}(h) = h^3 + 3\alpha h + 3(\sigma^2 + \alpha^2) + \varepsilon + 3\sigma^2\alpha + \alpha^3 + 12\varphi H$$

which is just $h^3 + 12\varphi H$ in the case of smooth bearings.

The concerned boundary conditions are

$$P = 0\,, \qquad x = -L \quad \text{and} \quad x = 0$$

Introducing the dimensionless quantities

$$x^* = \frac{x}{L}\,,\;\; P^* = \frac{h_0^3}{U\eta L^2}P\,,\;\; \mu^* = \frac{k\bar{\mu}\mu_0 h_0^3}{U\eta}\,,\;\; \varepsilon^* = \frac{\varepsilon}{h_0^3},\;\; \sigma^* = \frac{\sigma}{h_0}\,,\;\; \alpha^* = \frac{\alpha}{h_0}, \qquad \varphi^* = \frac{12\varphi H}{h_0^3}$$

$$A_1 = \varepsilon^* + 3\,\sigma^{*2}\alpha^* + \alpha^{*3},\; A_2 = 12\varphi^*,\; A_3 = 3(\sigma^{*2} + \alpha^{*2}),\; A_4 = 3\,\alpha^* \qquad (1)$$

and fixing the following symbols;

$$Q_1 = 1 + A_1 + A_2 + A_3 + A_4\,;\;\; Q_2 = 1 + \frac{A_4}{2} - \frac{A_1}{2} - \frac{A_2}{2}\,;\; a_1 = \sqrt{\frac{Q_1}{Q_2}}\,;\;\; a_2 = \sqrt{Q_1 Q_2}\,;\;\; C = \frac{k_1}{2Q_2}$$

$$k_1 = -\frac{\tan^{-1}(a_1\pi)}{a_2\pi}\;;\;\; k_2 = \left[-\left(\frac{1}{a_2} + \frac{1}{2Q_2 a_1}\right)\frac{\tan^{-1}(a_1\pi)}{\pi} + \frac{1}{2Q_2}\right];\;\; A = \frac{k_2}{a_2\pi} - \frac{k_1}{a_2\pi} - \frac{k_1}{2Q_2\pi a_1}$$

Substitution, one obtains the expression for pressure distribution in dimensionless form as

$$P^* = \frac{\mu^* \sin(\pi x^*)}{2} - \;[A\tan^{-1}(a_1\pi x^*) + C(x^*)] \qquad (2)$$

The dimensionless load carrying capacity per unit width is given by,

$$W^* = \frac{{h_0}^2}{U\eta L^3}\, W$$

$$= \int_{-L}^{0} P^*\, dx$$

As a result of which ,

$$= \frac{\mu^*}{6} - \left[A\left(\frac{1}{2a_1\pi}\ln\,|1 + {a_1}^2\pi^2| - \tan^{-1}(a_1\pi)\right) - \frac{C}{2}\right] \qquad (3)$$

The total friction F is,

Now friction

$$= L\int_{-L}^{0} \eta\left(\frac{\partial u}{\partial z}\right)\, dx$$

where

$$\left(\frac{\partial u}{\partial z}\right) = \frac{\partial p}{\eta\partial x}\left(z - \frac{h}{2}\right) + \frac{U_1 - U_2}{h}$$

The friction force is needed on the two surfaces z=h and z=0.Writting F_h for the first term and F_o for the second,

$$\frac{F_{h,0}}{L} = \int_{-L}^{0}\left[\pm\frac{\partial}{\partial x}\left(P - \frac{\mu_0\bar{\mu}H^2}{2}\right)\frac{h}{2} + \frac{(U_1 - U_2)\eta}{h}\right]\, dx \qquad (4)$$

It is shown that the upper pad is titled there will be a horizontal component of force, which exactly equals the difference in frictional drag between the top and bottom surfaces; if this term is added to the friction of the upper surface F_h it cancels the minus sign. The details can be seen from Cameron (1972). It gives directly non-dimensional friction,

$$F^* = \frac{1}{U\eta L} F$$

$$= \frac{\mu^*}{4} + \frac{A(1-4a_1^2)}{2a_1^2}\tan^{-1}(a_1\pi) - \frac{C}{\pi} \tag{5}$$

3. Results and Discussion

It is clearly seen that equation (2) determines the dimensionless pressure distribution; while the distribution of load carrying capacity in non-dimensional form is given by equation (3).Besides, the non-dimensional friction is given by equation(5). It is examined from Equation (2) and (3) that the increase in the dimensionless pressure and load carrying capacity as compared to the case of conventional lubricant respectively, turns out to be $(0.5\mu^* \sin(\pi x^*))$ and $(0.16)\mu^*$. The performance of the corresponding magnetic fluid based smooth bearing system can be obtained by taking the roughness parameters to be zero. Further, in the absence of magnetization this study reduces to the investigation carried out in Cameron (1972). Figures (2)-(5) describing the variation of load carrying capacity with respect to μ^* for various values of σ^* , ε^*, α^* and $\varphi *$ indicate that the load carrying capacity increases as considerably due to magnetic fluid lubricant, this increases being relatively less for the case of α^*.

The profile of the variation of the load carrying capacity with respect to σ^* for different values of ε^*, α^* and $\varphi *$ respectively are presented in Figures (6)-(8).It is clear that the effect of standard deviation is considerably adverse, in the sense that the load carrying capacity decreases significantly. The combined effect of $\varepsilon^* and\ \alpha^*$ on the distribution of load carrying is depicted in Figure (9). It is observed that positively skewed roughness decreases the load carrying capacity, while the load carrying capacity increases sharply due to negative skewed roughness. Identically, (+ve) α^* decreases the load carrying capacity while the load carrying capacity gets increased due to α^* (-ve). We have the distribution of load carrying capacity with respect to different values of $\varepsilon^*$and α^* to porosity in Figures (10)-(11). These Figures point out that the combine effect of roughness parameters and porosity parameter is considerably adverse.

Figures (12)—(21) present the variation of non-dimensional friction with respect to magnetization parameter, roughness parameters and porosity. It is observed that positive variance and standard deviation decreases the friction, while magnetization and negatively skewed roughness increases the friction. It is also found that the porosity decreases the load carrying capacity as well as friction. Further, this decrease is more in the case of variance. Some of the Figures suggest that the negative effect of standard deviation can be minimized by the positive effect of magnetization in the case of the negatively skewed roughness especially, when negative variance occurs.

4. Conclusion

This study suggests that the magnetization leads to overall improved performance of the bearing system. This article establishes that the bearing can support a load even when there is no flow. Further, this investigation makes it clear that the roughness must be accounted for while designing the bearing system from the life period point of view

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Nomenclature:

h	Fluid film thickness at any point	τ	Shear stress
F	Frictional force	σ	Standard deviation

H	Magnitude of the magnetic field	ε	Skewness
L	Length of the bearing	α	Variance
P	Lubricant pressure	φ	Porosity
U	Velocity	σ^*	Non-dimensional Standard deviation
W	Load carrying capacity	ε^*	Non-dimensional Skewness
h_0	Fluid film thickness at $x = 0$	α^*	Non-dimensional Variance
F^*	Dimensionless frictional force	φ^*	Non-dimensional Porosity
P^*	Dimensionless pressure	μ^*	Non-dimensional magnetization parameter
W^*	Dimensionless load carrying capacity	$\bar{\mu}$	Magnetic susceptibility
η	Dynamic viscosity	μ_0	Permeability of the free space

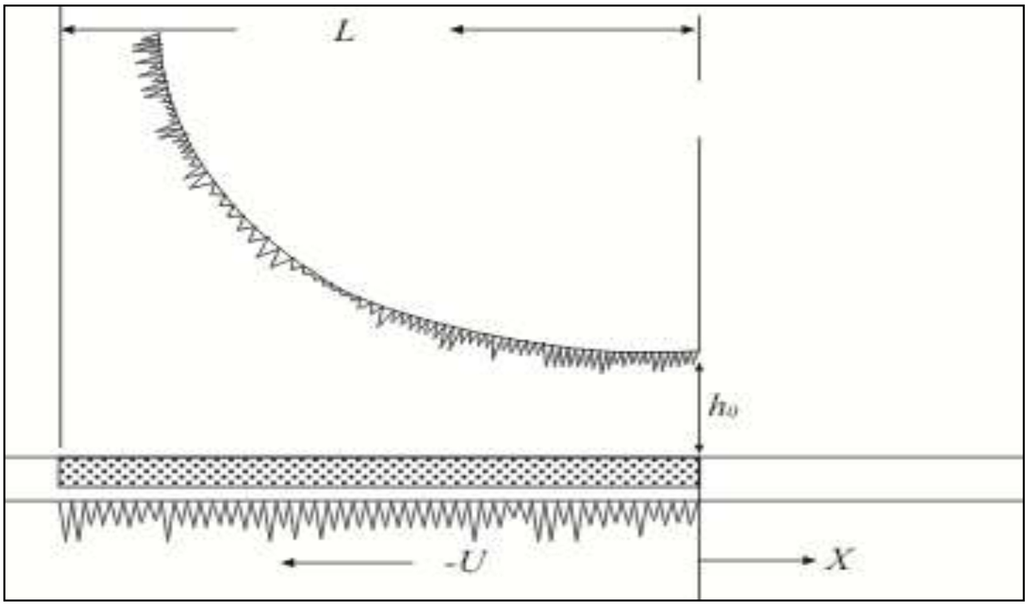

Figure 1. The Configuration of the Bearing System

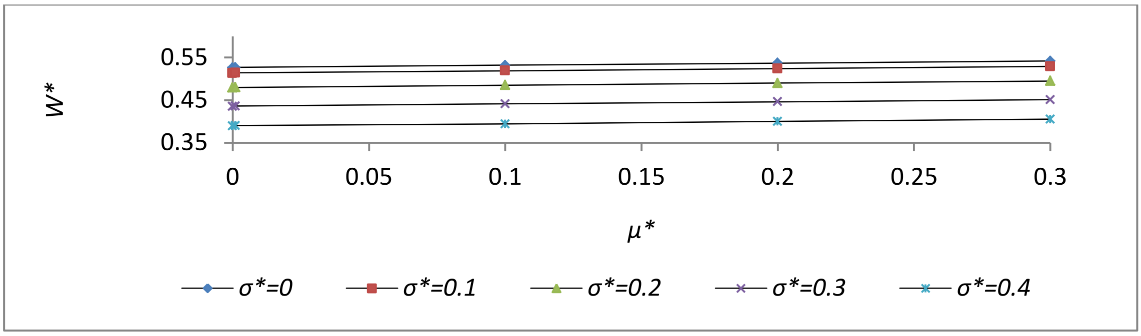

Figure 2. The Variation of Load carrying capacity with Respect to μ^* and σ^*

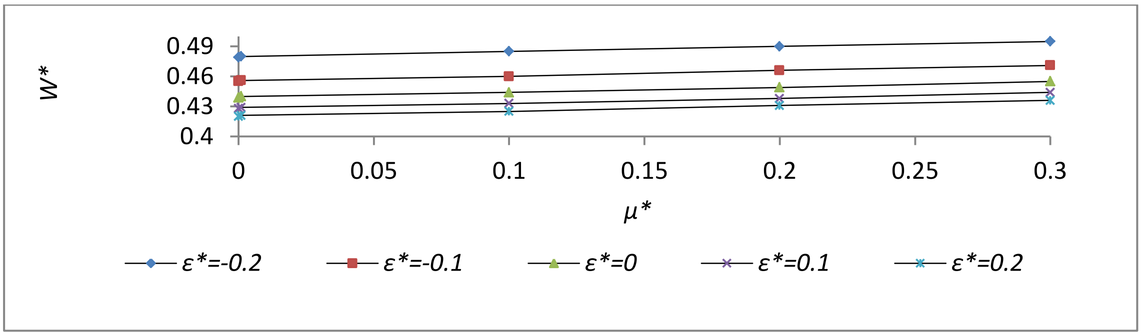

Figure 3. The Variation of Load carrying capacity with Respect to μ^* and ε^*

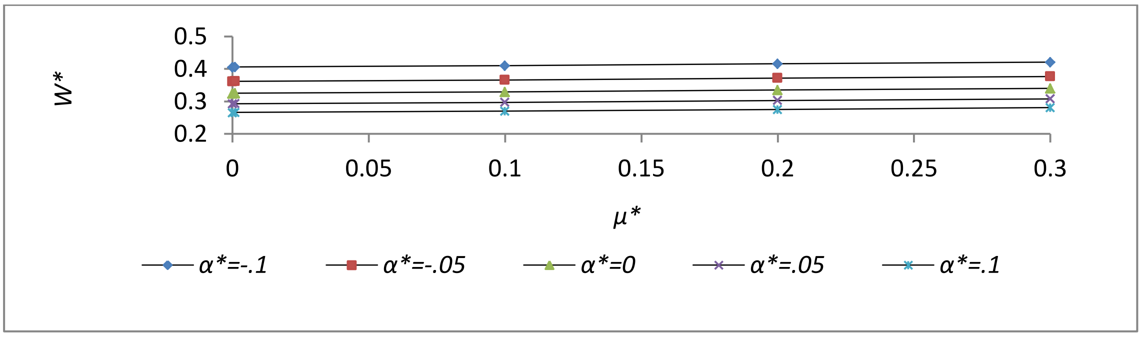

Figure 4. The Variation of Load carrying capacity with Respect to μ^* and α^*

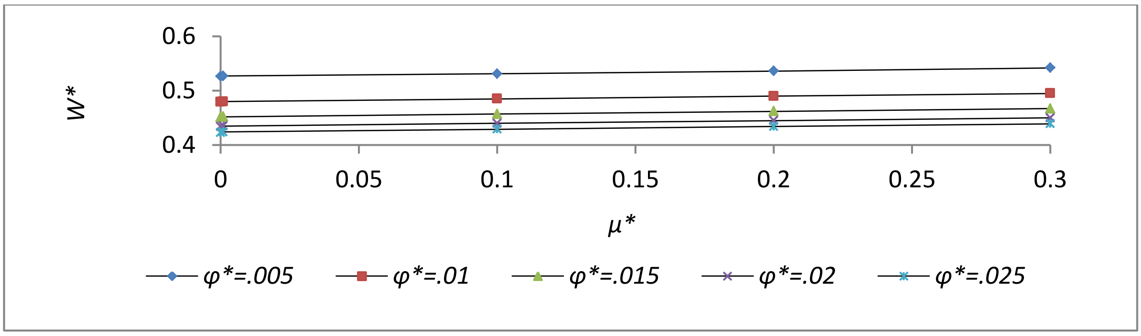

Figure 5. The Variation of Load carrying capacity with Respect to μ^* and φ^*

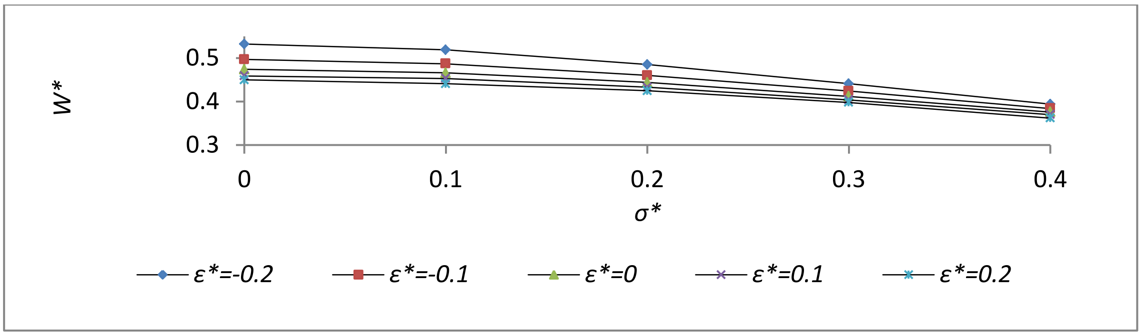

Figure 6. The Variation of Load carrying capacity with Respect to σ^* and ε^*

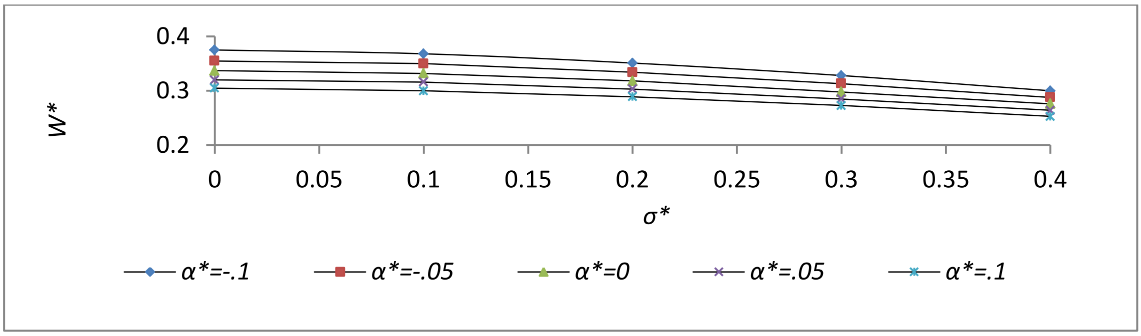

Figure 7. The Variation of Load carrying capacity with Respect to σ^* and α^*

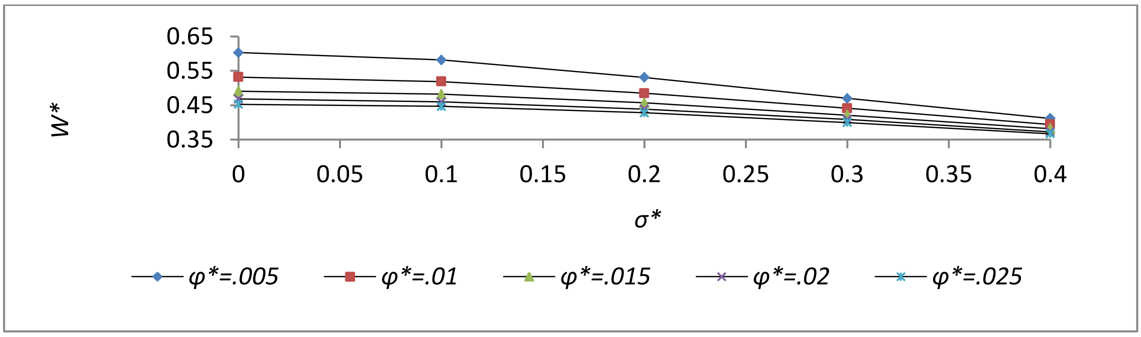

Figure 8. The Variation of Load carrying capacity with Respect to σ^* and φ^*

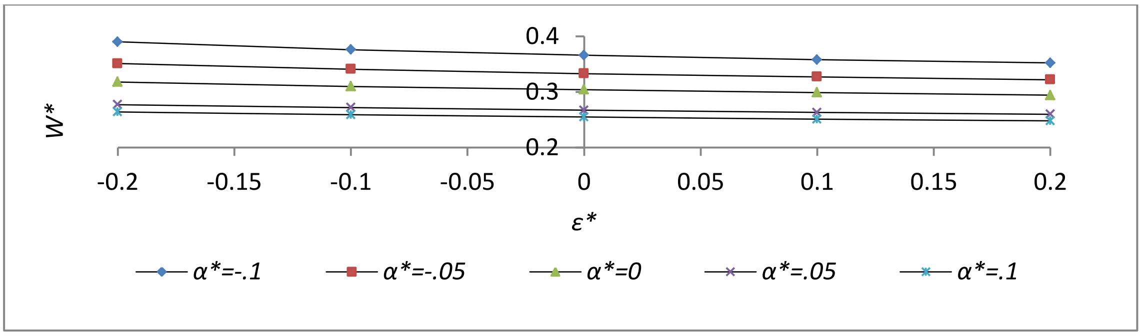

Figure 9. The Variation of Load carrying capacity with Respect to ε^* and α^*

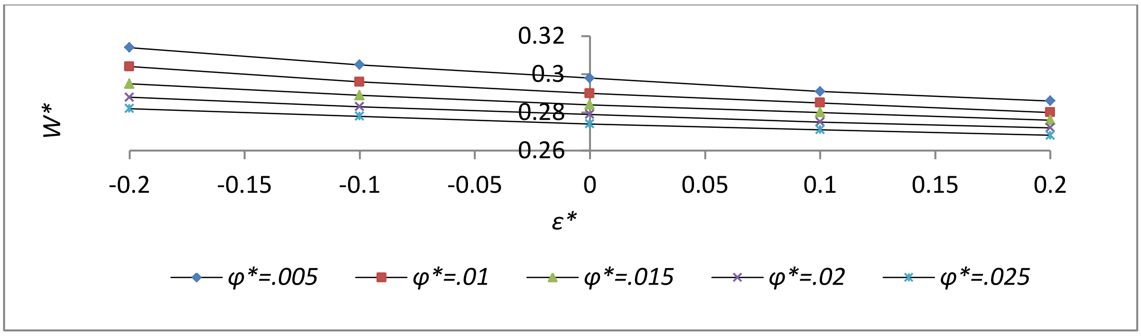

Figure 10. The Variation of Load carrying capacity with Respect to ε^* and φ^*

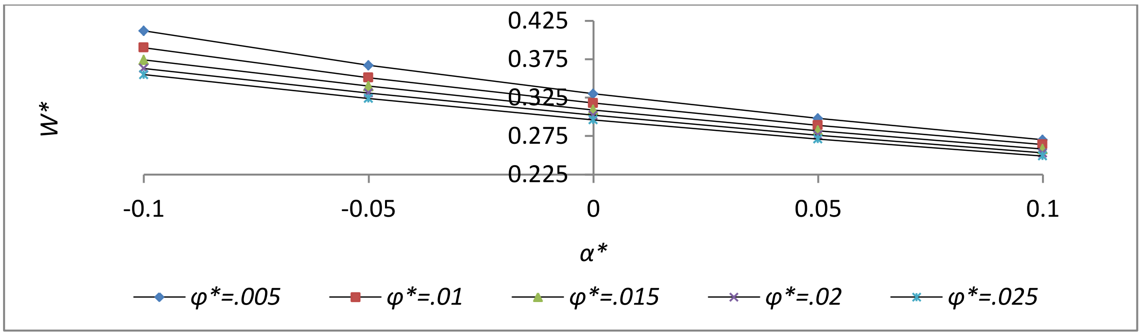

Figure 11. The Variation of Load carrying capacity with Respect to α^* and φ^*

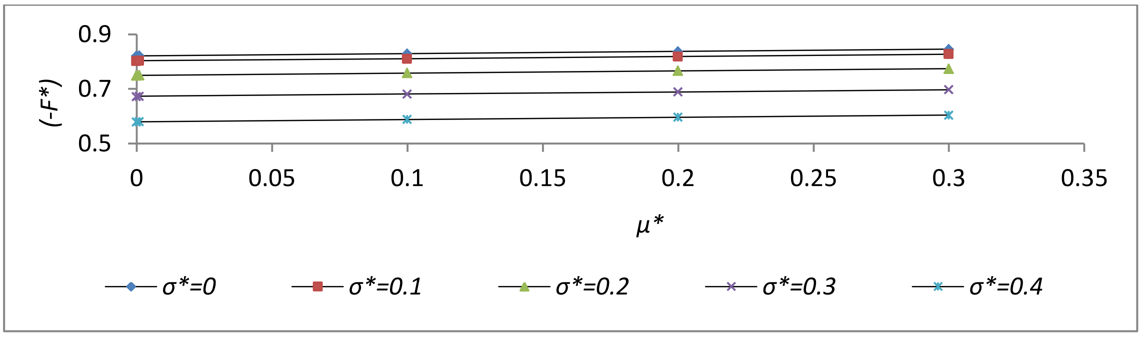

Figure 12. The Variation of Friction with Respect to μ^* and σ^*

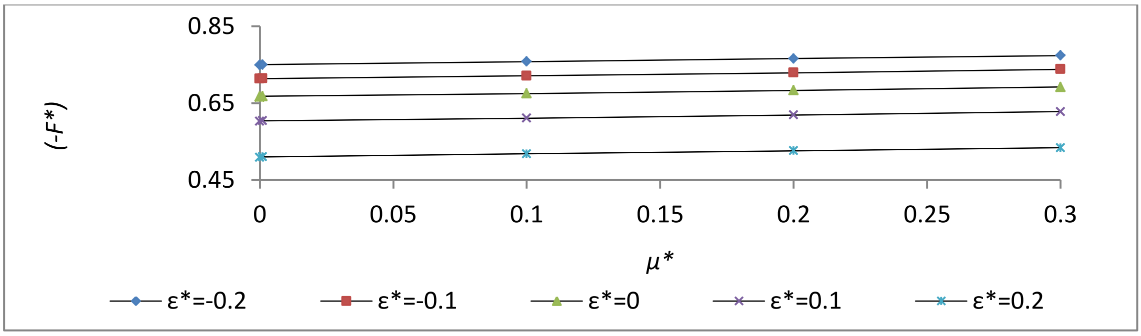

Figure 13. The Variation of Friction with Respect to μ^* and ε^*

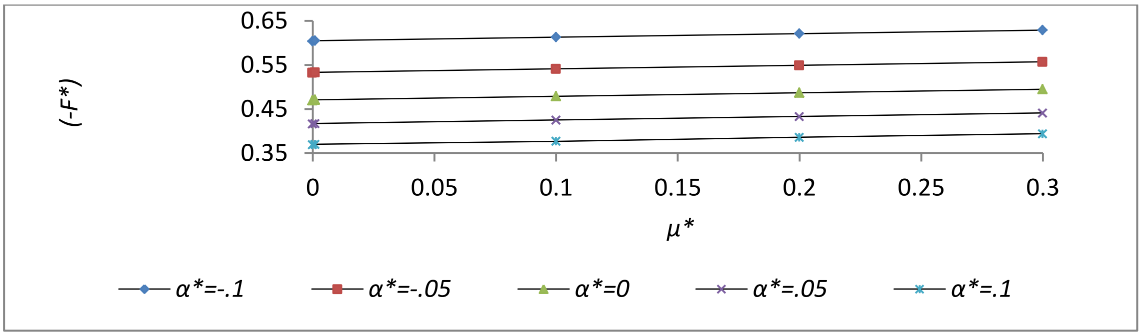

Figure 14. The Variation of Friction with Respect to μ^* and α^*

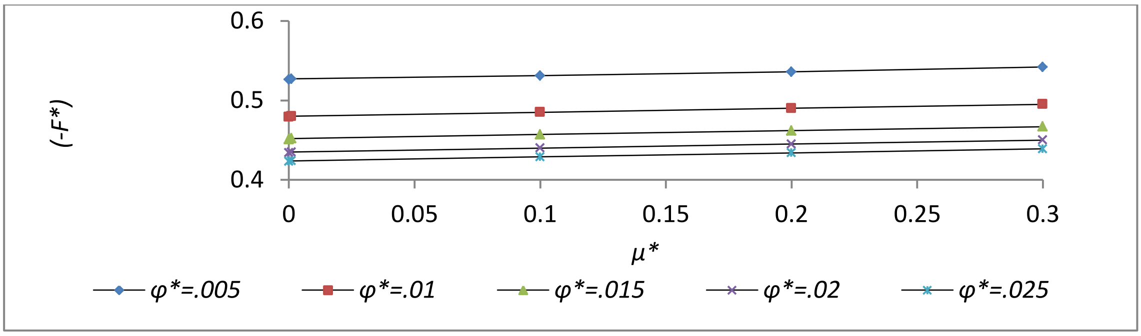

Figure 15. The Variation of Friction with Respect to μ^* and φ^*

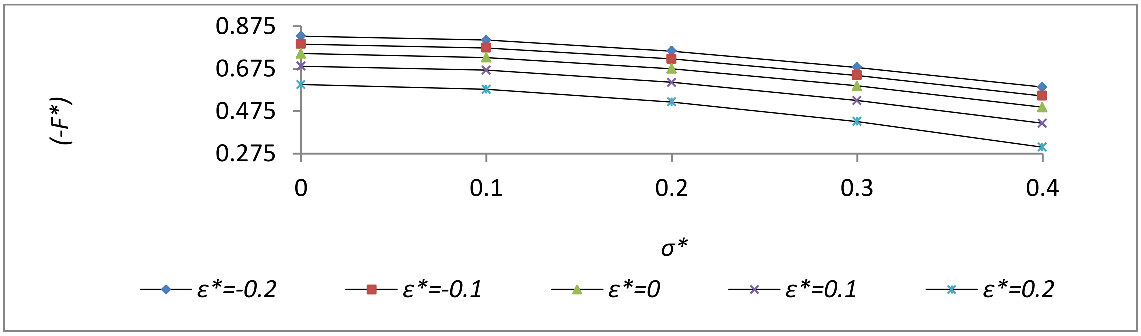

Figure 16. The Variation of Friction with Respect to σ^* and ε^*

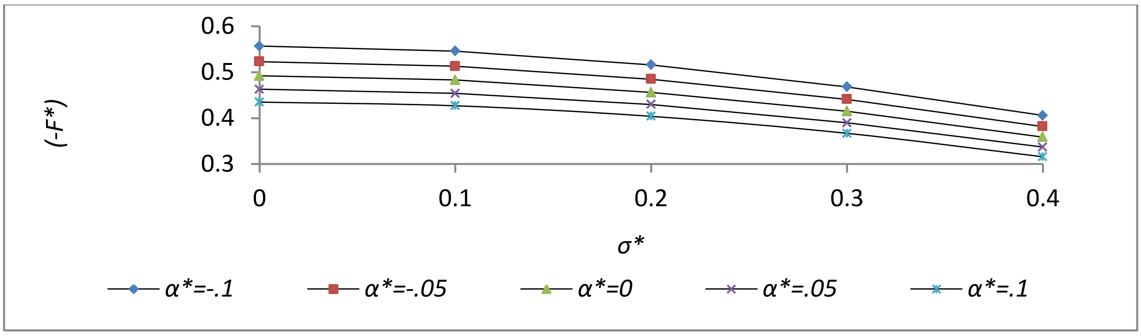

Figure 17. The Variation of Friction with Respect to σ^* and α^*

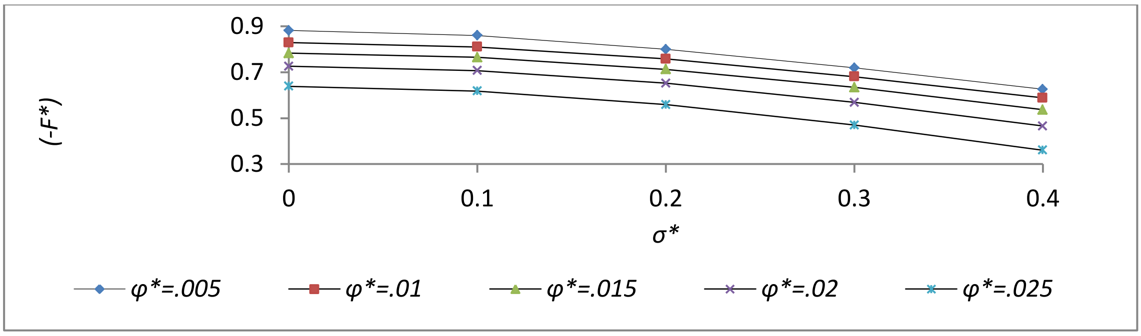

Figure 18. The Variation of Friction with Respect to σ^* and φ^*

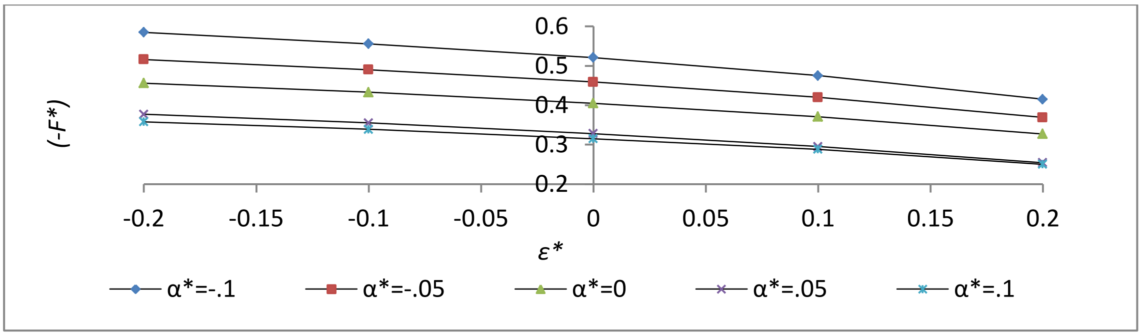

Figure 19. The Variation of Friction with Respect to ε^* and α^*

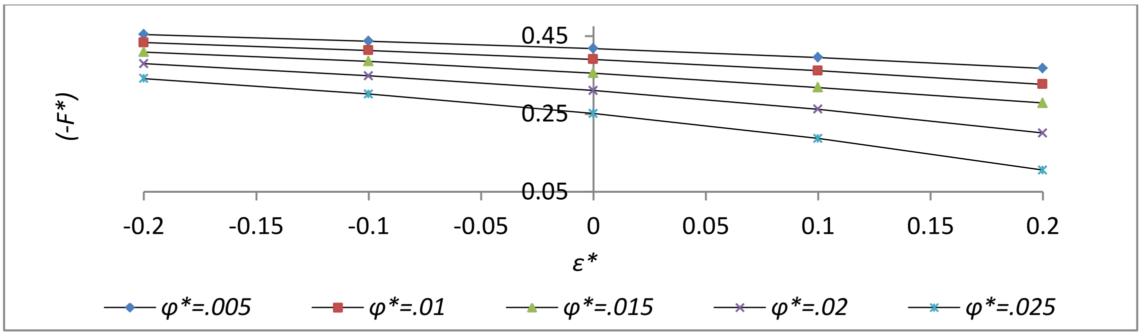

Figure 20. The Variation of Friction with Respect to ε^* and φ^*

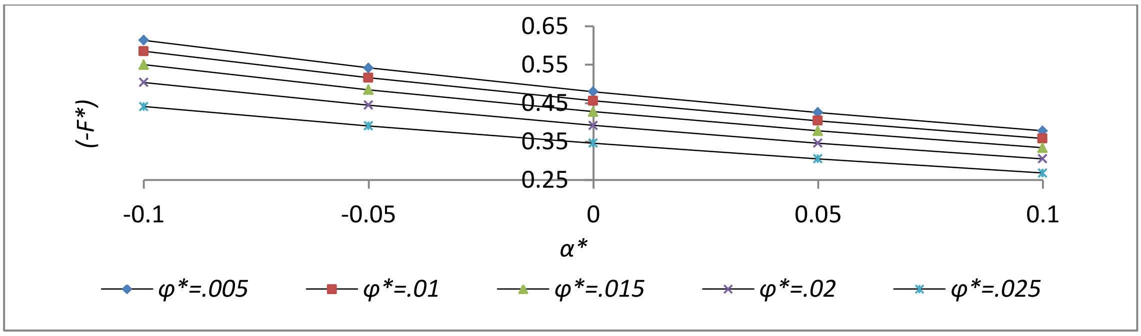

Figure 21. The Variation of Friction with Respect to α^* and φ^*